Experimental Evaluation of Position Control Methods for Hydraulic Systems

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***Abstract*—This paper presents a unified and systematic assessment of ten position control strategies for a hydraulic servo system with single-ended cylinder driven by a proportional directional control valve. We aim at identifying those methods that achieve better tracking, have a low sensitivity to system uncertainties, and offer a good balance between development effort and end results. A formal approach for solving this problem relies on several practical metrics, which will be introduced herein. Their choice is important, as the comparison results between controllers can vary significantly, depending on the selected criterion. Apart from the quantitative assessment, we also raise aspects which are difficult to quantify, but which must stay in attention when considering the position control problem for this class of hydraulic servo systems.**

***Index Terms*—Adaptive control, hydraulic systems, optimal control, proportional control, variable structure systems.**

I. INTRODUCTION

AUTOMATION of heavy-duty manipulators generated over the years considerable interest in low-level position control of a typical hydraulically actuated axis. The dynamic of hydraulic systems is nonlinear. Main contributing factors are the flow phenomena, oil compliance, and last but not least, friction in the actuators, especially in the case of cylinders. The last decade saw a number of results reported in modeling and control of machines with hydraulic actuation: modeling and identification [1], tracking control [2], control in the presence of friction [3], force feedback control [4], and impedance control [5]. On a broader plan, several control algorithms, such as robust [6], adaptive [7], and variable structure with sliding modes [8], have been implemented and studied in the general class of hydraulic servo-systems. With such a vast array of options, it becomes difficult to decide which method to implement in a practical application. The aim of this paper is to provide an overview of the main results obtained with different controllers for the benefit of the person responsible for implementing the controllers. We focused on controllers which are most often used in robotic control, but inevitably, the chosen set is far from being comprehensive.

This paper is arranged as follows. The description of the experimental test rig and the experiment design issues are presented in Section II, followed by a presentation of the various control methods and the reasons for which they were selected for investigation in Section III. The evaluation process implies the selection of several metrics, which are presented in Section IV. The quantitative assessment results are presented in Section V, while in Section VI we discuss some qualitative assessment issues. The main conclusions are summarized in Section VII.

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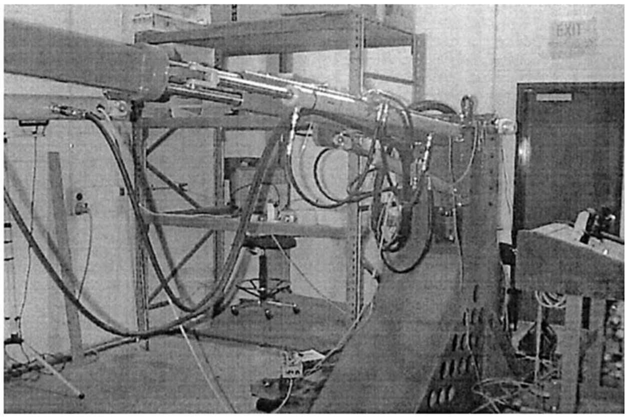


Fig. 1. The mining manipulator used as a test rig.

II. EXPERIMENTAL SETUP

A. System Configuration

The test-bed used is part of a four-degree-of-freedom (DOF) generic machine having the mechanical structure and the functional capability of the existing mining manipulators used in rock breaking and roof bolting operations. Standard off the shelf commercial components have been used in order to preserve the resemblance with the machines currently operating underground. The only difference is the replacement of the electrohydraulic on–off valves with proportional technology, and the addition of pressure and displacement sensors. The test rig is shown in Fig. 1.

The experiments reported in this paper were conducted on the pitch axis, consisting of a double acting, single-ended rod hydraulic cylinder (2.5″ $\times$ 1.5″) driven by a proportional directional control valve with a bandwidth of around 6 Hz. Connecting them are two 3/8″ hydraulic hoses, each having a length of approximately 6.5 m. This is one of the main characteristics of mobile machinery used in the mining and construction industries which puts additional burden on the controllers. Pressures at both ports are measured using typical transducers, while piston position is measured by an internal LVDT. All controllers were run at a rate of 50 Hz. An additional inline suppressor was installed at the valve "P" port to attenuate the supply pressure ripples.

B. Experiment Design

For each of the analyzed controllers, two types of experiments were conducted. The aim was to test the control techniques investigated for a variety of demands.

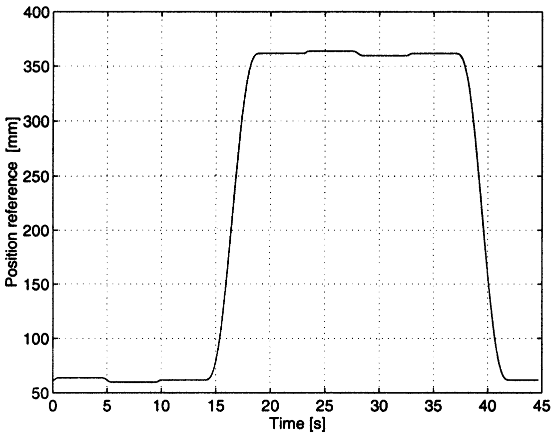


Fig. 2. Reference signal for point-to-point control.

In the first set of experiments the controller was tuned using a discrete-time sinusoidal reference signal of the form

$$y_{ref}(k) = 100 \cdot \sin(2\pi 0.1 kT) \qquad [\text{mm}] \tag{1}$$

with the sampling time $T = 0.02$ s. The amplitude of the signal insured that tracking would cover more than two thirds of the maximum piston stroke, and therefore would force the controller to deal with the nonlinearities induced by the continuous variation of the volumes of oil in the two cylinder chambers. In addition, such a signal generates a range of velocities from zero to the maximum achievable in each direction, and forces motion reversal. Most problems generated by friction occur in the low-velocity domain and when reversing the piston motion.

In the second set of experiments, a reference signal y_{ref} (shown in Fig. 2) was computed by a point-to-point trajectory generator based on a quintic polynomial. The controllers were run in this case using the parameters tuned for sinusoidal tracking. Apart from involving large and small displacements and velocities, the second set of experiments was also designed with the intent of testing the regulator capabilities of the controller in question. In the nominal case, the load was 1000 N, and the supply pressure set at 100 bar. To test the sensitivity to parameter variation, we conducted experiments in which the supply pressure was decreased to 50 bar, and the load had a periodic step-wise variation between the level 0–1000 N, with a frequency of 0.1 Hz.

III. Position Control Methods

This section introduces the methods chosen to solve the position control problem for the hydraulic servo system. The controllers in the set are as follows:

1) proportional derivative (PD);
2) acceleration feedback using an experimentally identified friction model (FRID);
3) acceleration feedback using a variable structure friction observer (VSO);
4) variable structure with sliding mode (VSC);
5) model reference adaptive control (MRAC);
6) self-tuning using a recursive least square parameter estimator (ST1);
7) self-tuning using a Kalman filter for parameter estimation (ST2);
8) pole placement (PP);
9) linear quadratic Gaussian (LQGC);
10) self-tuning generalized predictive control (GPC).

Rightfully dubbed the workhorse of automatic control, proportional-integral-derivative (PID) is usually the handy choice. The method is mainly attractive because it does not require a model, making it the fastest to implement. A PD version was implemented, given that the integral action is already present in the cylinder dynamics [9].

Friction is a major disturbance in hydraulic cylinders, and compensating it could improve the positioning performance. Conventional friction compensators applied in electric drives cannot be used in hydraulic systems, given the presence of complex nonlinearities. Compensation is achieved in such systems by tracking an acceleration reference signal, with the friction information being used via an acceleration estimate provided by an observer. Two friction observers are compared, one based on an experimental friction model (FRID), and another one based on a variable structure method (VSO) [10].

The VSC design is detailed in [9]. A fuzzy technique reported in [11] was used in order to minimize chattering and to determine the control in the boundary layer neighboring the switching surface. We previously reported on the application of MRAC for hydraulic servo position control [12]. PP was investigated partly as a prerequisite for self-tuning control, and partly because it is an established control design method.

Optimal methods have been included in order to analyze the position control problem in a stochastic framework. LQGC is more than often the benchmark test for optimal control methods, while GPC, which originated from the area of process control, is increasingly popular in the robotic control community. Research results have shown that GPC can handle unstable and nonminimum phase plants with unknown delays, and offers a certain degree of robustness. As far as hydraulic actuators are concerned, applications of the method were reported for position control in hydraulic motors [13] and force control in cylinders [14].

IV. Quantitative Measures for Controller Assessment

Linear control theory provides simple performance metrics which give indication of the command following capabilities in time domain. Unfortunately, some of the most common performance metrics are meaningless when comparing nonlinear and linear control methods, as they imply a (sometimes low-order) linearization of the plant. Therefore, their use here has been avoided, and a more practical approach was followed.

A. Some Auxiliary Variables for Metric Definitions

To denote the two different type of position reference signals applied to the controller, we will use a "string" variable $\mathcal{R}$, $\mathcal{R} \in \{\text{SIN, PTP}\}$ where SIN is the sinusoidal reference signal, and PTP is the point-to-point positioning task. This variable will be simply referred to as the "reference signal." The plant parameters will be $\mathcal{P}$, $\mathcal{P} \in \{\text{NOM, VAR}\}$, where $\mathcal{P} = \text{NOM}$ denotes the nominal plant parameters, while $\mathcal{P} = \text{VAR}$ denotes changes to parameter values from their nominal levels, as mentioned in Section II-B.

The metrics will be considered over a time interval $\mathcal{T}$ defined by the limits t_s and t_f

$$\mathcal{T} = [t_s,\, t_f] \tag{2}$$

where

$$t_s = \begin{cases} 0, & \text{if initial transients are included,} \\ 10s, & \text{if initial transients are not included.} \end{cases} \tag{3}$$

Using these auxiliary variables, the definition of various metrics used in evaluating the performance of the investigated controllers can be formalized in a convenient manner.

B. Selection of Metrics

For the position control problem, the controller accuracy would be naturally described by some metric involving the position error $e(t)$. The choice here was to consider an average value as well as the upper bound of the absolute errors. Keeping in mind the type of demands and the nature of the plant, the *mean* and *absolute positioning accuracy* (MPA and APA) are introduced first.

Definition 1: The MPA of a controller is defined as the root mean squared position error obtained for a reference signal $\mathcal{R}$, a plant condition $\mathcal{P}$, and averaged over a defined time interval $\mathcal{T}$

$$\text{MPA}(\mathcal{R},\, \mathcal{P},\, \mathcal{T}) = \left[\frac{1}{t_f - t_s}\int_{t_s}^{t_f} e^2\, d\tau\right]^{1/2}. \tag{4}$$

In view of the discrete nature of the signals, the integral can be numerically approximated using the trapezoidal method.

Definition 2: The APA of a controller is defined as the maximum absolute position error obtained for a reference signal $\mathcal{R}$, a plant condition $\mathcal{P}$, over a defined time interval $\mathcal{T}$

$$\text{APA}(\mathcal{R},\, \mathcal{P},\, \mathcal{T}) = \max_{\tau \in \mathcal{T}}\{|e|\}. \tag{5}$$

While MPA and APA give a direct indication of the positioning performance, they do not mirror the "effort" made by the controller in achieving it. In LQGC design for example, the cost function penalizes the output error and the control effort at the same time. The same idea is used here to define a weighted positioning accuracy (WPA), which measures the control activity and the associated position error.

Definition 3: The WPA of a controller is defined for a reference signal $\mathcal{R}$, a plant condition $\mathcal{P}$, and averaged over a defined time interval $\mathcal{T}$ in the form

$$\text{WPA}(\mathcal{R},\, \mathcal{P},\, \mathcal{T}) = \left[\frac{1}{t_f - t_s}\int_{t_s}^{t_f} [e^2 + \rho u^2] d\tau\right]^{1/2} \tag{6}$$

where $\rho > 0$ is a control weighting factor.

The integration is again computed numerically using the trapezoidal method. To give physical sense to the addition under the integral, the control weighting has the dimension $\langle$ m/V $\rangle$ in saturation index (SAT) units. A hint pointing to the right value of ρ comes from the practical implementation of the LQGC. Alternatively, an appropriate value can be determined such that $\sqrt{\rho}\, u(t)$ has the same order of magnitude with $e(t)$. The value used in our experiments was $\rho = 0.002$.

A practical and convenient way to assess the robustness of the controllers is to look at the positioning accuracy obtained for any of the two different reference signals with the nominal ($\mathcal{P} =$ NOM) and changed ($\mathcal{P} =$ VAR) plant parameters.

Definition 4: The robustness index (RI) of a controller is defined for a reference signal $\mathcal{R}$ and represents the relative error of the mean positioning accuracy MPA for nominal and changed plant parameters over the entire motion duration $\mathcal{T}$

$$\text{RI}(\mathcal{R},\, \mathcal{T}) = \frac{|\text{MPA}(\mathcal{R},\, \text{NOM},\, \mathcal{T}) - \text{MPA}(\mathcal{R},\, \text{VAR},\, \mathcal{T})|}{\text{MPA}(\mathcal{R},\, \text{NOM},\, \mathcal{T})}. \tag{7}$$

One of the potential dangers facing controllers associated with the hydraulic system is saturation. For a common reference signal however, some of the investigated control methods produced saturated control output, while others kept the signal between the admissible values. An SAT is introduced in order to quantify saturation.

Definition 5: The SAT of a controller is defined for a reference signal $\mathcal{R}$ and a time interval $\mathcal{T}$ as the proportion of the respective time interval during which the controller output is saturated

$$\text{SAT}(\mathcal{R},\, \mathcal{T}) = \frac{t_{\text{sat}}}{t_f - t_s}. \tag{8}$$

The control output is considered saturated when the computed control is $|u(t)| > 9$ V.

In discrete-time, and for a constant sampling rate, the index can be computed as

$$\text{SAT}(\mathcal{R}) = \frac{N_{\text{sat}}}{N} \tag{9}$$

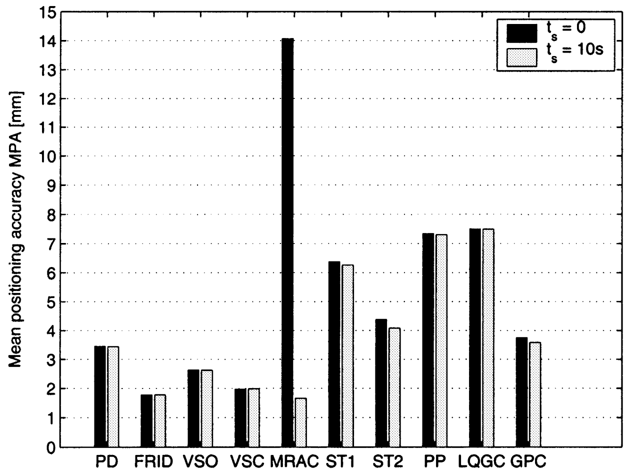
where N_{sat} represents the number of saturated control output samples from a total of N samples.

The ideal controller should comply with several requirements at the same time: it should minimize the output error with a minimum of effort, in spite of disturbances of different nature being present in the system. From this viewpoint, it is therefore useful to have a composite index (CI) based on all or some of the metrics introduced above.

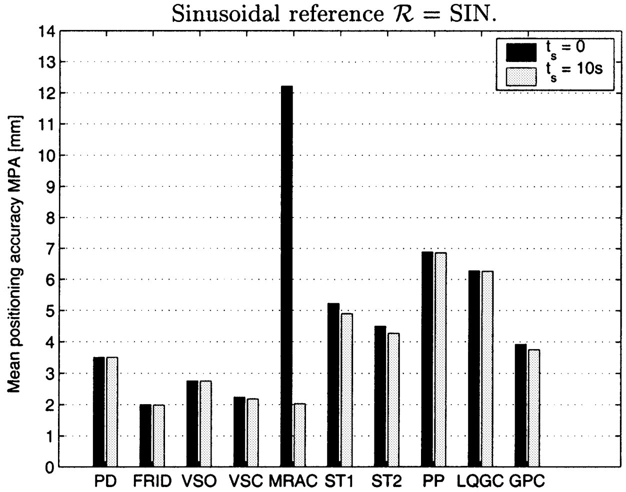
Definition 6: The CI of a controller is defined over a time interval $\mathcal{T}$ for a reference signal $\mathcal{R}$ as the weighted sum of the robustness index, and the absolute and weighted accuracies corresponding to the nominal plant $\mathcal{P} =$ NOM

$$\text{CI}(\mathcal{R},\, \mathcal{T}) = \sum_{\mathcal{R}}\{k_1\text{RI}(\mathcal{R},\, \mathcal{T}) + k_2[\text{APA}(\mathcal{R},\, \text{NOM},\, \mathcal{T}) + \text{WPA}(\mathcal{R},\, \text{NOM},\, \mathcal{T})]\}. \tag{10}$$

Note in the definition of CI, that a higher emphasis is placed on the accuracy and the robustness properties of each of the controllers, but the control effort is still in balance. The saturation index SAT is less relevant, as the saturation effects are already mirrored by the weighted position accuracy WPA, while the computation index was discarded for reasons already mentioned. The values chosen for the weighting factors were $k_1 =$

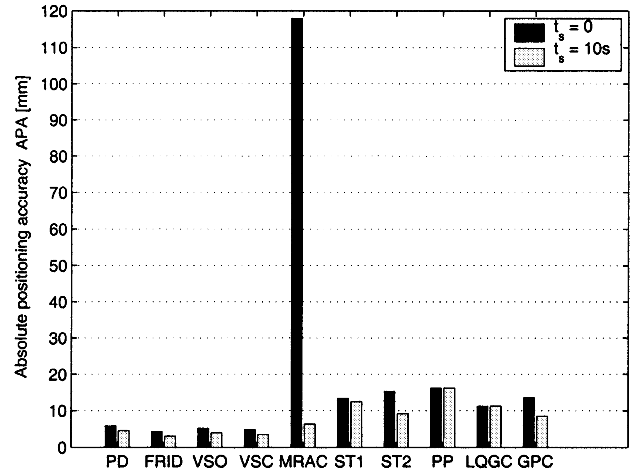


Sinusoidal reference $\mathcal{R}$ = SIN.

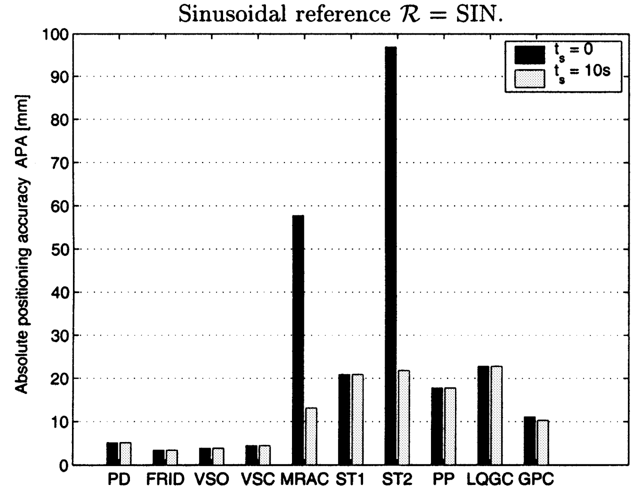


Point–to–point positioning $\mathcal{R}$ = PTP.

Fig. 3. Mean positioning accuracy MPA of various controllers, for nominal plant $\mathcal{P}$ = NOM.



Sinusoidal reference $\mathcal{R}$ = SIN.



Point–to–point positioning $\mathcal{R}$ = PTP.

Fig. 4. APA of various controllers for nominal plant $\mathcal{P}$ = NOM.

10 and $k_2 = 1/2$. The presence of these factors in (10) could attract some criticism as there seems to be an arbitrary selection process of their values. Also, APA and WPA have length units, while RI is dimensionless. From a numerical point of view however, the "best" controller minimizes each of the metrics APA, WPA, and RI. Given the linear combination of these measures in (10), the relative comparison using the CI will reward the accuracy for higher values of k_2 and robustness for higher values of k_1. The values suggested for comparison purposes here are $k_1 = 2$, and $k_2 = 1$.

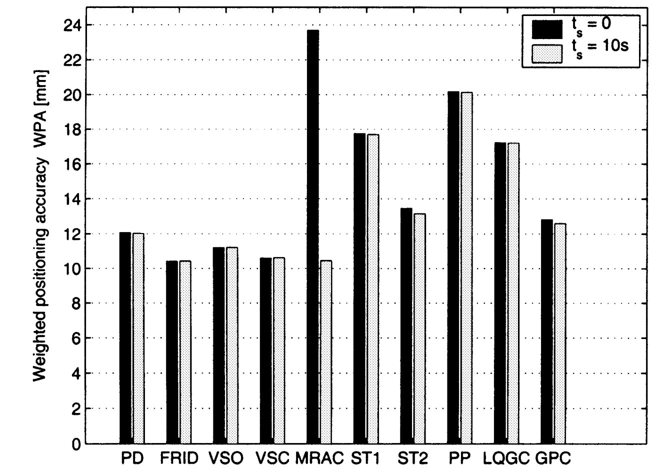
V. Quantitative Assessment Results

Using the metrics defined in the Section IV, each controller in the selected set will be assessed with respect to its suitability for position control in hydraulic servo systems.

The MPA was computed for each controller in the set, for the nominal plant $\mathcal{P}$ = NOM, and for both types of reference signals $\mathcal{R}$, leading to the results plotted in Fig. 3. The influence of the initial transients for both demands is considerable in the case of MRAC, some minor differences being noted for the controllers using online identification of plant parameters, namely ST1, ST2, and GPC. The significant difference in the case of MRAC is explained by the lack of prior information concerning the controller parameters, which bear no direct relation to plant parameters. Excluding initial transients however, the controller achieves the best mean positioning accuracy, followed closely by FRID, VSC, and VSO. Notably, the controllers directly using linear representation of the system achieve the worst accuracy. There is some benefit in providing online parameter estimation in ST1, ST2, and GPC, compared with PP and LQGC which are based on fixed parameters. PD lies in the middle of the scale.

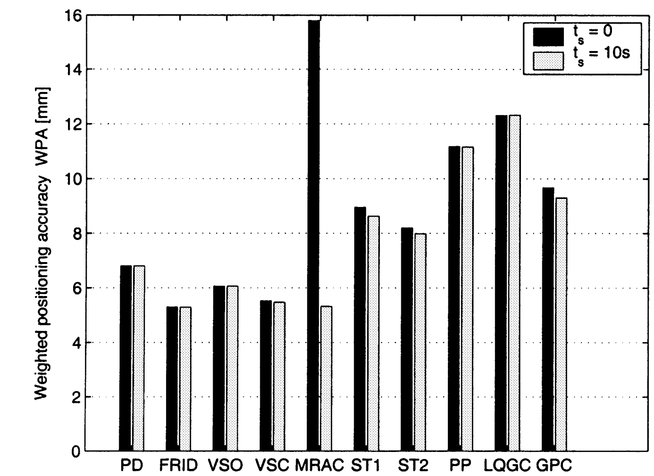
In terms of the absolute positioning accuracy APA, the results shown in Fig. 4 are favorable to FRID, VSO, VSC, and PD. In spite of neglecting the initial adaptation transients, maximum errors place MRAC behind them, while the controllers based on linear process models are again at disadvantage. No major changes are observed when penalizing the control effort in addition to the position error. All the observations made earlier regarding the mean accuracy in Fig. 3, are valid for the WPA shown in Fig. 5.

Some predictable results were obtained with the RI plotted in Fig. 6. The most robust controllers were VSC and GPC, as expected. Acceleration feedback with both of the observers, MRAC and PD, displayed average robustness properties. It



Sinusoidal reference $\mathcal{R} = \mathrm{SIN}$.

(a)



Point–to–point positioning $\mathcal{R} = \mathrm{PTP}$.

(b)

Fig. 5. WPA of various controllers for nominal plant $\mathcal{P} = \mathrm{NOM}$.

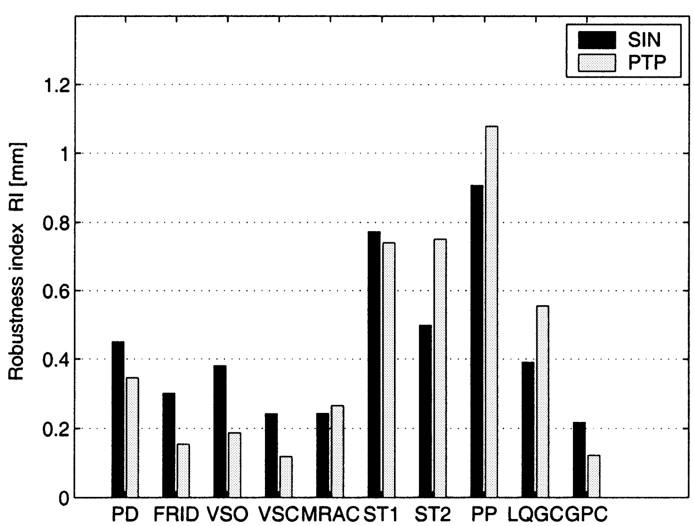
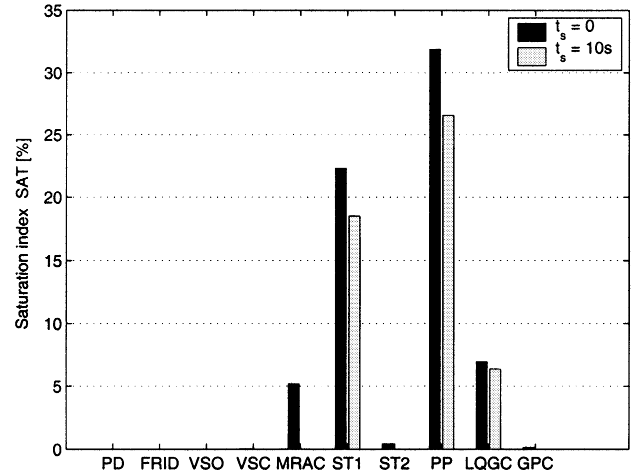


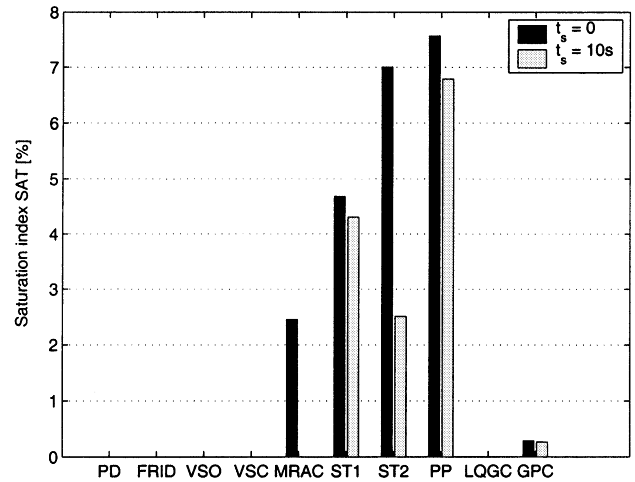
Fig. 6. RI for different reference signals $\mathcal{R}$.

was also confirmed that self-tuning is not a suitable method for control when plant parameters change suddenly, something that



Sinusoidal reference $\mathcal{R} = \mathrm{SIN}$.

(a)



Point–to–point positioning $\mathcal{R} = \mathrm{PTP}$.

(b)

Fig. 7. SAT of various controllers, for nominal plant $\mathcal{P} = \mathrm{NOM}$.

was expected in theory from MRAC as well. Methods which are based on fixed plant parameters, i.e., PP and LQGC, had obviously problems in dealing with the changes.

Although saturation effects are reflected in the accuracy indexes, it is worthwhile having a look at the SAT, shown in Fig. 7. Saturation seems to be present in general in controllers designed with explicit use of a linear plant model. Notable exceptions are the optimal control methods, where the cost function penalizes the magnitude of the control output. The sinusoidal reference signal imposes higher demands than the point-to-point positioning, and as a result leads to higher saturation index values.

Finally, the composite performance index produced the results listed in Tables I and II, for $t_s = 0$, and $t_s = 10$ s with weighting factors $k_1 = 2$ and $k_2 = 1$. The ranking of the controllers in both tables is almost identical, with the exception of MRAC, which achieves the lowest score if initial transients are taken into account, and the second highest when these transients are discarded. Recall that the main difficulty in MRAC is to find suitable initial values for the controller

TABLE I
RANKING OF CONTROLLERS BASED ON THE CI, FOR $\mathcal{R}$ = PTP, $k_1 = 2, k_2 = 1$

$t_s = 0$ s		
Rank	Controller	CI [-]
1	FRID	16.62
2	VSC	16.85
3	VSO	18.41
4	PD	20.45
5	GPC	21.18
6	ST2	24.77
7	ST1	30.69
8	LQGC	31.45
9	PP	34.37
10	MRAC	40.68

TABLE II
RANKING OF CONTROLLERS BASED ON THE CI, FOR $\mathcal{R}$ = PTP, $k_1 = 2, k_2 = 1, t_s = 10$ s

$t_s = 0$ s		
Rank	Controller	CI [-]
1	FRID	16.62
2	MRAC	16.80
3	VSC	16.81
4	VSO	18.40
5	PD	20.01
6	GPC	20.58
7	ST2	24.17
8	ST1	30.30
9	LQGC	31.42
10	PP	34.32

parameters. One solution is to run the system first with null initial conditions in the estimator, and once the convergence of the parameters is achieved, use these values to initialize the estimator in subsequent runs. The best performance is achieved in order by FRID, VSC, and VSO, with the amendment for the second place in the case of $t_s = 10$ s as noted above. A valid alternative is also PD. The fact that PP was worse than LQGC deserves an explanation. Three issues are at stake in PP design: the identified model, the choice of the controller poles, and the choice of the observer pole. The identified model had an ARX structure, and no attempt was made to characterize the noise in the system. In LQGC, the identified model included a representation of the noise, and there was less flexibility in assigning the controller and observer poles. They were derived straight from the model.

The influence of two other pairs of weighting factors (k_1, k_2) on the CI is shown in Fig. 8. If the composite index rewards the robustness properties more than the achieved accuracy, by increasing k_1 and/or decreasing k_2, the changes at the top of the list are not significant.

VI. QUALITATIVE ASSESSMENT

Some of the possible assessment criteria carry a higher degree of subjectivity than others, and for this reason it is extremely difficult to quantify them in a meaningful way. The most eloquent example is the modeling effort required by the design of a control algorithm. If we consider the extremes, then the complexity of the modeling phase is the highest for VSC, LQGC, GPC, and

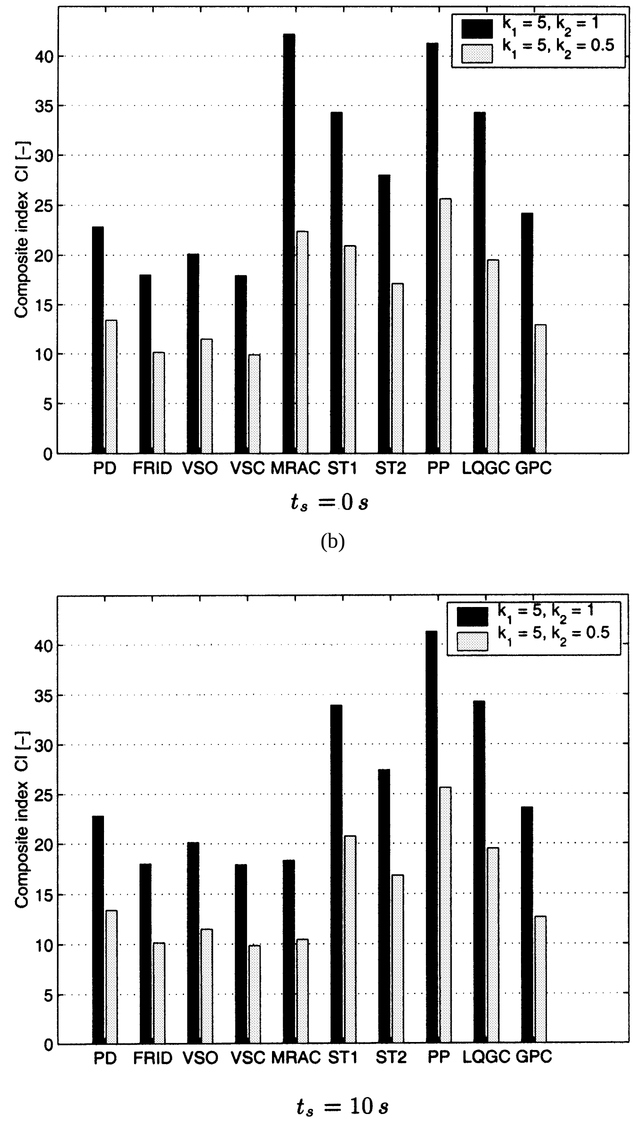


Fig. 8. Influence of the weighting factors on the composite index CI.

FRID, and the lowest for PD. VSC required a higher physical insight into the process than other methods studied. For GPC and LQGC on the other hand we used a black-box system model with difficulties arising in the formulation and validation of the noise model. The experimental friction identification is troublesome, considering the many factors that influence friction, and in addition FRID requires information on external forces.

Another important criterion is the number of sensors required. The major factors which advocate for keeping their number to a minimum are price, limited reliability, and increased hardware complexity. In addition, there are instances where sensors cannot be located in optimal positions, for various reasons. The acceleration feedback controllers and the VSC normally require four sensors per DOF, for measuring the piston position, the pressures at the cylinder ports, and the external force acting on the piston. All the other controllers work with position information only. Note, however, that in the experimental setup used, the hydraulic cylinder has a built-in position transducer. In general, retrofitting a position sensor to a heavy-duty machine is an expensive exercise, especially for long-stroke and large-diameter cylinders. String potentiometers seem to be the solution of choice, but they are less accurate and reliable than the built–in type.

Due to the applicative nature of this research, practical implementation issues play an important role, and among them, the ease of controller tuning can be considered relevant. From this viewpoint, the ideal controller should be characterized by as many as possible of the following properties:

- online self-adjusting capabilities; this is clearly the case in the adaptive controllers, but autotuning of PID controllers is also achievable;
- low number of parameters to adjust; in the PP case for example, no parameter adjustment is required, but given the limited performance, it is not a good choice for exemplification; a better one would be MRAC, where only the adaptation gain needs to be adjusted; at the opposite scale, VSO has the highest number of tuning parameters;
- the existence of clear guidelines for parameter setting; in most of the cases, tuning was based on a combination of limited guidelines and a trial-and-error approach; the exceptions are PD, and MRAC, where starting values were taken from results reported by other authors investigating similar systems.

VII. Conclusion

This paper has provided an experimental evaluation of some of the most common position control algorithms for hydraulically actuated equipment typically used in field robots. Two of the closed-loop system properties were targeted, namely tracking and robustness. Given the diversity of the control structures investigated, it was important to formulate adequate assessment criteria. Several quantitative measures were used, with the main ones based on position error.

Overall, the best performance was achieved with FRID, followed closely by MRAC (with properly initialized parameters) and VSC, with the difference between the later two being insignificant in practical terms. Worthwhile mentioning is also VSO, which then confirms that acceleration feedback with friction estimators in the feedback loop is a strong contender for the controller of choice. Average performance was obtained with PD, with the other analyzed controllers lagging behind. When considering the robustness properties alone, the order is altered by the GPC, which takes the first spot. All controllers had very similar computational demands, with the exception of GPC which was nearly twice the average, but incorporated on-line parameter estimation.

The quantitative measures alone fail in giving the complete dimensions of the selection process. A qualitative assessment was made involving issues which are difficult to be formulated mathematically. They related mainly to the modeling effort required by the design of a certain control algorithm, and the type and number of sensors required by each control paradigm.

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